TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 869

THE CONVERSION OF ENERGY IN A RADIATOR

By A. Weise

Gesammelte Vorträge der Hauptversammlung 1937 der
Lilienthal-Gesellschaft für Luftfahrtforschung
vorgetragen in München vom 12-14, October 1937

Washington
July 1938

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 869

THE CONVERSION OF ENERGY IN A RADIATOR*

By A. Weise

INTRODUCTION

The importance of the radiator has grown with the technical development of the airplane: as the engine power increases the amount of heat that must be removed grows in importance. The aerodynamic refinements on the airplane itself have made the share of the radiator on the power loss stand out more and more. The advances in high-speed flight had the same effect, since the power quota for the same installation needed to pull the radiator through the air increases as the square of the speed.

It was therefore a worthwhile task for the research to find ways to lower the cooling drag. And the success of these efforts has been so great that without this research the modern high-speed airplane would have decidedly inferior performances. These results were the outcome of painstaking exploration into the physical processes accompanying the cooling in airplane flight.

Stratosphere flying has now given the cooling problem a new motive. On the one hand, the amount of heat that must be dissipated is greater, and this applies not only to the cooling medium of the engine but also to the supercharging air and in given cases to the exhaust gas used in the turbine. Moreover, the decreasing air density renders the transfer of heat difficult, especially in the stratosphere where the outer air temperature no longer drops, that is, can no longer act against the density effect. The result is that the radiator air is heated very considerably in relation to its absolute inlet temperature. This has far-reaching results for the behavior of the radiator, which previously had been neglected on ground radiators.

The treatment of modern cooling problems calls for the coordination of thermodynamics and aerodynamics and, for sonic velocities, the kinetic theory of gases.

It is the purpose of this report to give a comprehensive discussion of all the fundamental principles and physical phenomena that offer a key to the understanding and the solution of modern cooling problems.

Behavior of Radiator Core

The design of the modern radiator lends itself to a division of the problems into behavior of the actual heat exchanger, the "radiator core," and its combined action with the airplane, as defined by the "installation."

The radiator core is an exchanger of heat designed with special regard to minimum weight. The exhibits a, b, and d of figure 1 show that the basic element of the radiator cores on the side of the cooling air is the straight tube of, in general, constant section. A number of such tubes are combined to form a honeycomb-like system. The heat taken from the cooling medium is passed on to the walls of these tubes, partly direct by scavenging, partly indirect through heat conduction, that is, fin effect. The physical conditions on the side of the cooling medium are of subordinate importance. Even the finned tube radiator of figure 1(c) may be freely considered as radiator with straight tubes. Thus, the treatment of heat exchange in the radiator is reduced to the conventional problem of heat exchange in a tube.

The radiator core mounted in an airplane acts in its installation no differently than when mounted in a closed channel on the torque stand, where it is studied for its cooling capacity and pressure losses.

Consider then the radiator block under these conditions (fig. 2). Assume for the present that the radiator is charged with cold air rather than a hot liquid, so that the velocity of the air before and behind the radiator is the same. The energy loss of the passing air due to friction manifests itself as pressure drop that can be measured with a differential pressure manometer fitted before and aft of the radiator. Now, if the radiator is used as heat exchanger, the heating of the cooling air increases the

volume and, consequently, the velocity of the air along
its path through the cooling passages. One effect of the
increasing velocity is the increasing pressure drop due
to friction. A further pressure drop is caused by the
change in momentum associated with the acceleration of the
air, that is, by a mass effect.

The gas-dynamic pursuance of the friction process with
heat transfer in a tube indicates certain limits within
which the conditions in the tube must move and which are
governed by the initial condition and the initial velocity.
These limits are most conveniently presented in the pres-
sure-speed diagram, since both are of direct interest in
the further treatment.

Assume that point A in figure 3 corresponds to the
condition of the cooling air at entry into a heat-exchang-
ing tube. It (A) is determined by the corresponding adia-
batic curve and the pressure, or the speed. At friction
without heat exchange the further conditions in the tube
follow the well-known Fanno curve, whose physically poten-
tial conditions terminate on reaching sonic velocity (point
B), because beyond it the entropy would have to decrease
again. With heat input the conditions follow the Fanno
curve. The upper limit is formed by the tangent to the
adiabatic curve in the starting point of the phase. This
straight line is followed if the heat is transferred with-
out friction. It, too, has its terminal point G on reach-
ing sonic velocity. This end point and that of the Fanno
curve as well as of all intermediate phase curves have as
geometrical locus a straight line through the origin of the
coordinate system. The result is a closed area A B C, with-
in which all phases of the air flow during the process of
heat transfer in a tube must take place. The static tem-
perature of the air can rise or drop during the heat trans-
fer. On the upper straight boundary, it reaches a maximum
at the pressure equal to half the pressure cut off on the
ordinate axis by this straight line. The total temperature,
that is, the sum of static and dynamic temperature, natu-
rally continues to increase with heat input and reaches a
maximum at sonic velocity. This presentation of the phases
of the air will be employed later on.

First, we will introduce the fundamental physical con-
nection between pressure drop and heat transfer on a wall,
already recognized by Reynolds, and which affords consider-
able simplifications for the subsequent analysis.

Consider, to begin with, a turbulent flow; that is a
flow characterized by the fact that gas bubbles are thrown
back and forth transverse to the principal flow direction
between the gas layers of different speed. Now assume
that such a gas bubble has lost its speed on the hot wall
as a result of frictional shear stresses and has through
conduction taken on the temperature of the wall. At a
distance from the wall the gas (air, for example) has a
certain speed. Then, if the gas bubble flies through a
turbulent transverse motion into this layer the following
effects take place. First, there is an impact of the gas
masses of different forward speed on collision, followed
by loss of mechanical energy, that is, change into heat
energy. This energy conversion manifests itself as a pres-
sure loss. But another side of the process is the mixing
of the gas mass heated at the wall with the cold mass of
the nucleus, that is, the transport of the heat quantity
absorbed by the gas bubble at the wall into the flow. It
will be seen that heat transfer and pressure drop are sim-
ply different manifestations of one and the same physical
process, hence that there must be some law governing the
two processes.

This argument holds for turbulent as for laminar flow
processes as presented for instance, in the boundary layer.
The connection with the above argument is found if we ana-
lyze the individual gas molecules rather than the gas bub-
bles. According to the kinetic theory of gases the gas
temperature is given through the kinetic energy of the ir-
regular molecular motion. The pressure is the shock effect
of this motion on the walls, which depends upon the energy
as well as on the number of impinging molecules, that is,
the density. Aside from the irregular motion, the molecules
are superposed by the regulated motion of the flowing gas,
which is likewise converted on the wall by the decelerating
shock effect of the wall molecules into irregular motion,
that is, arrested. Molecules thus slowed up at the wall
then speed through the molecular erratic whirling velocity
into the layers farther away from the wall on which the ad-
vancing motion is superposed. Thither they carry along
their irregular heat motion which was increased on the wall
as a result of the contact with the hotter, that is, faster-
moving wall molecules due to impact. They transport heat.
But they also lower the whirling energy of the molecules
remote from the wall if entrained by them through shock in-
to the general forward motion. The result is a pressure
drop. Thus we find the same relationship as for the turbu-
lent motion.

The mathematical form of the presumed law is most ex-
pediently found by dimensional analysis of the effectual
physical quantities. They are: the transferred heat
quantity Q (kcal/kg), the pressure drop ΔP (kg/m²), the
specific heat at constant pressure c_p (kcal/kg°), the
temperature difference ΔT between wall and gas nucleus,
the gas density ρ (kg s²/m⁴) and the velocity w (m/s).
According to the dimensional analysis, the following ex-
pression must be constant:

$$\frac{\Delta P \; c_p \; \Delta T}{Q \; (\rho \; w) \; w} = \Omega = \text{constant}$$

(ρ w) being the invariable stream density in the tube of
constant section.

The constant Ω can, after closer analysis of the dif-
ferential equations of the speed and temperature, be fur-
ther defined. It is approximately equal to the Prandtl
number Pr for gas, (more exactly $\Omega = 0.65 + 0.35$ Pr),
that is, for air, approximately $= 1$. The formula shows
that the heat is transferred so much more beneficially,
i.e., with so much less expenditure of pressure drop, that
is, on energy, as the flow velocity is lower and the tem-
perature difference greater.

This holds, for the present, only for the heat trans-
mission through friction. On the practical heat exchangers
a divergence from this relationship is to be expected be-
cause of the concurrent fin effect and flow separation.
But a mathematical check on the numerous radiator blocks
measured in the closed channel indicates that the quoted
expression is a constant even for practically employed
blocks independent of rate of flow, depth of block and mean
temperature difference (fig. 4)., which, of course, con-
tinues to diverge upwardly from the theoretical constant
as the share of the pure wall friction on the heat transfer
becomes less. The smooth-tube radiator with directly scav-
enged cooling surface manifests an almost perfect accord
with the theoretical requirement, i.e., the constant is ap-
proximately equal to 1. For tubular radiators with indi-
rect cooling surface it lies higher (at around 1.5), since
part of the heat at lower temperature difference is trans-
ferred as a result of the heat conduction in the fins. The
water-tube radiators with large, indirect cooling surface

and disclosing flow separation at the tubes have a con-
stant almost twice as high as the theoretical value. On
these, in consequence, twice the energy must be expended
for the same amount of heat transfer. In return for it,
they have the advantage of less weight and less contraction
of air section, so that for equal speed before the frontal
surface the heat is still transferred at lower speed, that
is - as the formula indicated - more propitiously.

For the ensuing investigations it therefore necessi-
tates for each radiator block system only one single factor
which completely defines its behavior with regard to the
aerodynamic-thermodynamic energy conversion.

The equation deduced from the dimensional analysis for
a long tube can be integrated and thus affords the law be-
tween the friction-pressure drop ΔP_r, the pressure drop
due to momentum change ΔP_b, the logarithmic mean of the
absolute air temperature in the tube T_m, the logarithmic
mean temperature difference Δ_m between wall and air, and
lastly, the block constant Ω:

$$\frac{\Delta P_r}{\Delta P_b} = \Omega \frac{T_m}{\Delta_m}$$

This relation also holds for intensive air heating, but not
if approaching sonic velocity. It affords everything need-
ed for the further radiator theory. The complete dynamical
gas theorem, which retains its validity even in the range of
velocity of sound, affords a differential equation which is
practical for following the phases in the tube for given
wall temperature and also facilitates its inclusion into
the pressure-speed diagram.

This brings us to the combined effect of radiator and
airplane.

Installed Radiator

The radiator installation has the purpose of render-
ing the rate of flow through the radiator block, which, on
the installed radiator, appears as effect of the flying
speed, controllable, and of keeping the external flow re-
sistances small, especially preventing separation of air

flow at the exposed surfaces of the radiator block.

To achieve the last aim, the radiator must be enveloped by walls that adapt themselves to the streamlines conditioned by the radiator and the other parts of the airplane. Figure 5 illustrates such an installation ventrally on a fuselage or engine nacelle. The cowling is so designed that the streamlines of the cooling air on entry to and exit from the cowling are parallel, thus assuring at both points the pressure of undisturbed flow. This need not be that way, as we will show later on.

The principal parts of the radiator installation are the diffuser and the nozzle. The first creates a divergent flow before the block, that is, raises the pressure conformally to Bernoulli's equation. This increase in pressure serves to overcome the pressure drop in the radiator block that - as is seen - in its installation finds itself under the same conditions as earlier in the closed channel. If the block has not absorbed the entire positive pressure produced in the diffuser a nozzle may, through convergence of the streamlines, reduce the air leaving the block to outside pressure.

One important factor for the functioning of the radiator is the rate of discharge, that is, the ratio of velocity before the radiator frontal surface w_k to the velocity of the undisturbed flow w_∞. It defines the positive pressure created in the diffuser. An energy loss within the channel is reflected in lower discharge than inflow velocity, that is, according to the equation of continuity, larger discharge section than entrance section.

In the determination of the radiator resistance, the external resistances due to skin friction on the outside of the cowling, due to separation of flow and interference with other parts of the airplane may be disregarded, since they fall under the general aerodynamics of the airplane. The remainder discussed here is termed "internal cooling resistance."

However, the transition from the consideration of energy loss in the block to the internal resistance of the installed radiator requires more than simply equating the internal energy loss defined by the pressure drop of the block, to the expendable energy which is defined through the radiator resistance and the flying speed. There are

other energy quantities; for instance there is a hidden
energy loss in the energy of the wake flow of the radiator,
the speed of which equals the difference between flying
speed and rate of discharge from the radiator. There also
is a thermodynamic gain due to the heat input at higher
pressure, in whose calculation the concurrently affected
discharge energy must be allowed for.

It is best to avoid the presentation of all separate
energies and to compute the internal radiator resistance
by applying the momentum theorem, which is especially sim-
ple, if it may be assumed that the pressure in the dis-
charge section is equal to the undisturbed initial pres-
sure. Then the integration of the momenta over the infin-
itely large flow section before and behind the radiator
gives as resistance the product of mass in time unit pass-
ing through the radiator and the wake velocity, that is, the
difference of undisturbed velocity and discharge velocity.

This method of determination affords a clear presenta-
tion in the pressure-speed diagram that retains its validity
even in the case of gas-dynamics, that is, even at high ve-
locities, because it not only affords the phases in the tube
in comprehensive form but also the processes of compression
in the diffuser and of the expansion in the nozzle which
either appear as adiabatic curves or - when allowing for
flow losses - as polytropic curves. The examples in figure
6 do not show these losses, since nozzle and diffuser can
be so designed that the deviations from the adiabatic curve
are small.

The starting point 1 of the compression on the adia-
batic curve for the pertinent air condition lies at flying
speed w_∞ (fig. 6). Following this adiabatic curve to the
air inflow velocity into the tube w_k immediately gives
the pressure increase in the tube. The discharge coeffi-
cient $\dfrac{w_k}{w_\infty}$ is plainly noticeable. The terminal point of
compression 2 is simultaneously the starting point of the
phase changes in the tube as previously discussed. This
process must take place within the previously defined
boundaries. If the radiator is without nozzle the pressure
drop in the tube continues up to the discharge pressure
from the radiator (point 3), which in the simplest case (as

in fig. 6) is equal to the initial pressure p_∞ . The case without nozzle is presented in figure 6(a), (c). If the radiator block is fitted with a nozzle, the block does not consume all of the pressure produced in the radiator. The pressure drop in the block lasts only as far as point 2', and the residual pressure gradient serves to accelerate the air in the nozzle. This speeding up process again follows an adiabatic curve which, once, as a result of the frictional pressure drop lies in the block and, then, owing to the air heating in the meantime, follows a course different from compression adiabatic curve.

Figures 6(a) and 6(b) present the case of the cold radiator. The compression from 1 to 2 is followed by the process in the block on the Fanno curve. Regardless of whether a nozzle is added (fig. 6(b)) or not (fig. 6(a)), the rate of flow through the radiator is lower than the flying speed. That is the radiator has resistance. The length 1-3 in the p-v diagram is a direct criterion for it. On the cold radiator with nozzle the resistance for equal coefficient of flow is smaller.

The compression in the diffuser does not change on the heated radiator if the coefficient of flow remains the same, but owing to the heat input in the tube, the phase curve there is to the right of the Fanno curve, which at first is noticeable on the radiator without nozzle (fig. 6(c)) in a greater discharge velocity, that is, lower resistance. If the previously cited boundaries for the phase changes are taken into account, it is found that the warm radiator, even if the friction is allowed for, may even have a negative resistance, that is, forward force. On the radiator with nozzle a flatter curve of the nozzle adiabatic in the range of higher temperatures can create a propulsive force even if the radiator block of itself because of excessive friction would have none (fig. 6(d)).

But this theoretical proof of potential radiator propulsion affords as yet no clear picture of the forces causing the propulsive force. It cannot originate in the radiator block itself because its force is opposite to the speed of flight, since frictional shear stresses only can attack its sides. Consequently, the radiator propulsion must be the resultant effect of forces on the sloping surfaces of the radiator cowling and, since - as indicated above - even a radiator without nozzle can have propulsive force, the diffuser must at least share in this phenomenon.

In the discussion of the mechanism of the origin of
the propulsive force, we first consider a radiator instal-
lation without block, that is, without friction. The air
in the diffuser is retarded and, as the initial pressure
is to prevail again at the nozzle outlet, accelerated by
the same amount in the nozzle. Rate of inflow and dis-
charge are alike. The resistance is zero. In spite of
that, forces come into play. The slowing up in the dif-
fuser effects an internal propulsive force, which, however,
is canceled by the identically great interference force of
the nozzle. On the radiator with friction, that is, with
installed block, the external pressure can be regained by
the friction effect even without nozzle. But then the
frictional stresses in the block produce interference forces
and these exceed on the cold radiator the propulsive force
of the nozzle, thus leaving a positive resistance. On the
warm radiator but without nozzle the block manifests – as
previously shown – aside from the friction, a further cause
of pressure drop, namely, the mass effect of the air on
accelerating following the volume increase as a result of
heating. This pressure drop makes it possible to regain
the outside pressure again without necessitating correspond-
ing wall forces in the block or in a nozzle, since the sup-
port occurs on the accelerated mass. The corresponding part
of the diffuser propulsive force is not compensated, that
is, remains effective. If the friction effects are small
enough it may lead to a positive propulsion. The warm ra-
diator with nozzle receives yet an additional propulsive
effect which fundamentally is created in the same manner.

The radiator cowling in figure 5 presents merely a
special case, namely, of parallel flow in the inlet and
outlet section. But it is not at all necessary and it
practically never happens that the flow there is parallel,
that is, the pressure equal to that of the undisturbed
flow.

The next discussion is on the effect of this deviation
on the energy conversion and the force effect on the radia-
tor.

Figure 7 compares several cases, that of figure 5 be-
ing shown as figure 7(a). The shaded area in the radiator
outlet indicates the mixing zone where the air layers of
the outside air moving at different speeds have exchanged
momentum with the air already passed through the radiator.
By this exchange of momentum, the mechanical energy is

transformed to heat energy. In spite of that, there is no additional resistance in this case because the total momentum remains constant during the mixing process.

Figure 7(b) shows the same radiator with shortened diffuser, that is, enlarged inlet section. The question is whether in this case the rate of flow changes and how the streamlines proceed. Since the conditions of flow at the radiator exit are as before determined by the form of the nozzle cover and the conditions within nozzle and block themselves are the same, the radiator requires as before the same velocity and the same pressure before the frontal area of the block. Now, is the shortened diffuser still in position to meet this requirement? It is, because the air before the inlet area in the free flow is dammed up as a result of which the streamlines themselves diverge as if the original diffuser length still existed. This streamline flow is made possible through the effect of the diffuser wall which then acts similarly to an airfoil. If the flow is the same as that on the unshortened diffuser, then the force effects must also be the same. Since only a part of the forces can appear on the inside of the diffuser, because of elimination of part of the wall area, the remaining part must be produced at the outside. This comes to pass through the flow around the front of the cover at increased velocity, that is, lower pressure. This negative pressure produces the part of the propulsive force of the diffuser formerly created on the inside. This increase of velocity also naturally increases the hazard of separation of flow on the outside, and must be counteracted by suitable design. According to test data the shortened diffuser in conjunction with high compression has a substantially better efficiency than the unshortened diffuser.

Figure 7(c) shows the shortened diffuser and shortened nozzle. The discharge here takes place at a pressure in excess of the undisturbed outside pressure. Here also the boundary streamline could follow the same course as if the wall existed, provided that, in constrast to the conditions on the diffuser as a result of the velocity difference between the air leaving the radiator and that flowing past at the outside, no mixing would be provoked again. It should be noted that the mixing in this case influences the inside cooling resistance, whose magnitude and sign depend upon whether the mixing occurs at positive or negative pressure, on the magnitude of the masses sharing on the mixing, and on the extent of the heating of the air in the radiator.

In general the undisturbed outside pressure is reached
again before any appreciable mixing of air masses takes
place, so that the mixing effect on the internal resist-
ance amounts to very little. The considerations regarding
the shortened diffuser also afford the key to the under-
standing of the control of a radiator. It was shown that
the size of the entrance section had no effect on the flow
since it is determined by the nozzle. It is therefore
possible also to regulate the flow of the radiator from the
nozzle without diffuser adjustment.

Figure 8 shows a radiator with three different control
settings. The diffuser is fixed while the nozzle in all
three settings on the outside is so designed that it does
not affect the flow on the exit due to the other parts of
the airplane; the air can always leave the radiator at the
same pressure. Thus the ratio of exit section of the noz-
zle and of the inlet section of an assumedly whole diffuser
is constant at all control settings. But, as the diffuser
inlet section does not change the course of the streamlines
before the diffuser must conform to the cited conditions.
Enlarging the nozzle to increase the flow (b) reduces the
divergence of the streamlines before the diffuser, that is,
the pressure rise diminishes. Ultimately the point is
reached where the diffuser acts as a whole diffuser, that
is, does no longer dam up in the free stream. On opening
the nozzle still further (c), the streamlines converge be-
fore the diffuser, which is now too long, or in other words,
has a too narrow entrance section, and negative pressure
prevails in the inlet. Then the diffuser must overcome
within a pressure rise greater than the effective pressure.
This, of course, entails unnecessarily large losses which
lower the flow relative to the diffuser with little loss
and increase the resistance. This behavior of the stream-
lines must be forestalled.

General Radiator Theory

From the foregoing arguments a general radiator theory
can be developed that can be used in conjunction with the
pressure-velocity diagram in the treatment of a radiator on
the airplane, even with gas-dynamical generalization, and in
answering all quantitative questions theoretically for any
heating, provided the velocities are not abnormally high.
On experimental bases only the cited constant of the radia-
tor block system containing the relation between heat trans-
fer and pressure drop is required. With this constant, the

pressure losses at any air heating can be directly assessed without having to resort to individual test data. One thereby eliminates the tedious repeated computations and first attains in fact the analytical solution of the problem. Having then computed the internal radiator resistance according to the previously described methods, the actual test is reduced to determining the radiator block constants and the external radiator resistances by the customary aerodynamic methods. The depth of the block itself must be so proportioned that the computed air heating or the computed pressure drop occurs. This theory supplies the answers to radiator problems so far as it relates to a combination of aerodynamics, thermodynamics, and gas dynamics. The results of such a theory are illustrated on a few examples.

Figure 9 gives for a radiator block constant $\Omega = 1$ the curve of the relative air heating in the radiator block η (ratio of temperature rise to initial temperature difference between air and wall) plotted against the coefficient of flow η_{ae} (here the ratio of velocity in the cooling channel to flying speed for a diffuser efficiency = 1) for different nondimensional initial temperature differences ξ (ratio of absolute cooling surface temperature to absolute air temperature at entry into the radiator block). $\xi = 1$ for the cold radiator, about $\xi = 1.2$ for the water radiator at sea level, $\xi = 1.8$ for the high-temperature liquid radiator in the stratosphere, and around $\xi = 5.5$ for an exhaust cooler in the stratosphere.

Figure 10 presents for $\Omega = 1$ the curve of the nondimensional resistance Ω_Θ of a radiator without nozzle plotted against the coefficient of flow. The nondimensional resistance is no resistance coefficient of the frontal radiator surface, but is at constant ξ directly proportional to the resistance* that the airplane must overcome

---

*
$$W = \Omega_\Theta \ \frac{Q \ w_\infty}{g \ c_p \ \Delta T}$$

where:

W   is resistance (kg)

Q,   heat dissipation (kcal/s)

w_∞, flying speed (m/s)

g,   gravity

c_p, specific heat of air

ΔT, initial temperature difference between wall and air

to carry off a certain amount of heat. It therefore af-
fords a better criterion than the drag coefficient, which
necessitates first the determination of the frontal area
of the radiator before the actual resistance is obtained.
The curves of the example in figure 10 are indicative of
a certain coefficient of flow at which the resistance be-
comes minimum. This fact, confirmed here in theory, has
been previously established in experiments. The explana-
tion for the minimum is that, at very high rates of flow,
the resistance becomes greater because the heat is carried
off at a high rate, that is, uneconomically, while at low
rates the greater heating of the cooling air consistently
reduces the mean temperature difference, and so lowers the
economy of heat removal.

Another general result of the radiator theory is as
follows: the power loss quota referred to engine output
increases at equal coefficient of flow and equal nozzle
setting at the square of the flying speed while the flying
height exerts an influence on the cooling resistance
through the absolute outside temperature but not through
the pressure or density. On the other hand, the variable
air density affects the size and consequently the weight
of the radiator.

These outlines of a general radiator theory should be
sufficient to indicate what steps the subsequent elabora-
tion of the fundamental concepts must follow to assure its
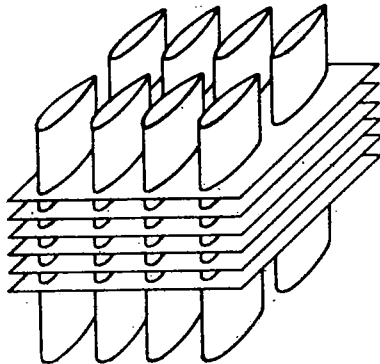practicability.

Translation by J. Vanier,
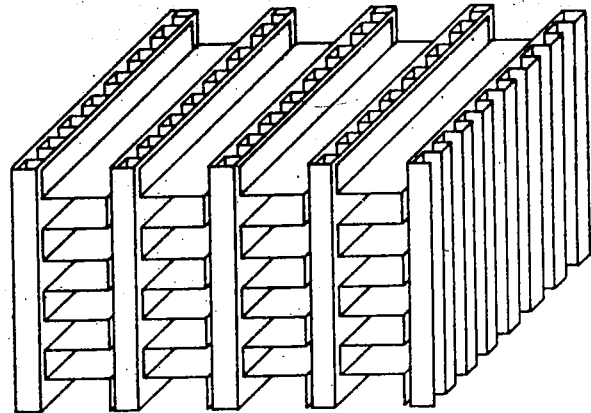National Advisory Committee
for Aeronautics.
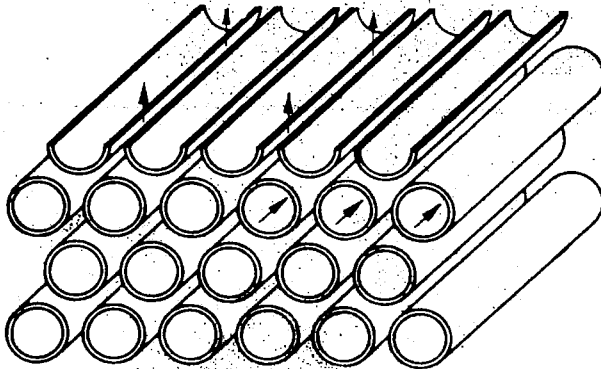
(a) Air-tube radiator without indirect
    cooling surface.
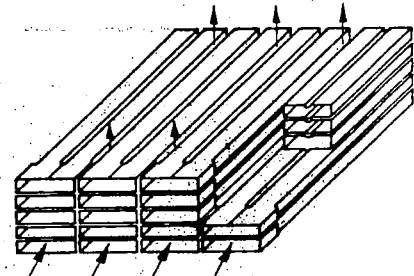
(b) Flat-tube radiator with
    indirect cooling surface.

(c) Finned-tube radiator.

(d) DVL-air radiator.

Figure 1.- Types of block radiators. (the cooling air is
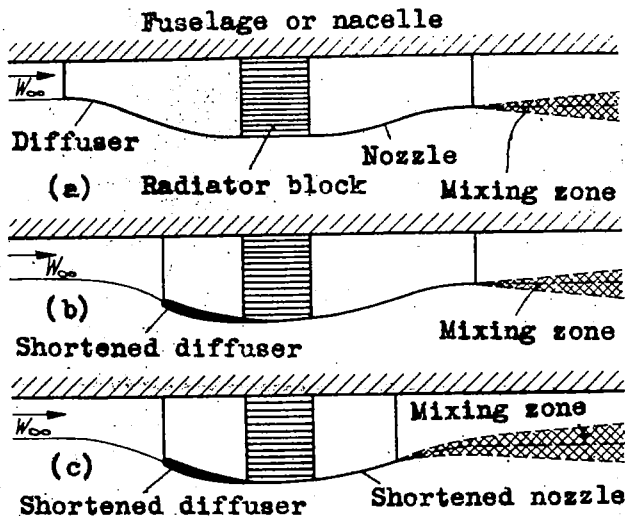   horizontal from front to back).

Fuselage or nacelle

W_∞

Diffuser   Nozzle

(a)   Radiator block   Mixing zone

W_∞

(b)
Shortened diffuser   Mixing zone

W_∞ Mixing zone

(c)
Shortened diffuser   Shortened nozzle

Figure 7.- Radiator installation.
            (change of flow following
    modified inlet and outlet section).

w_2 $\mathcal{T}_2$

a

Radiator block

Δp

w_1 $\mathcal{T}_1$
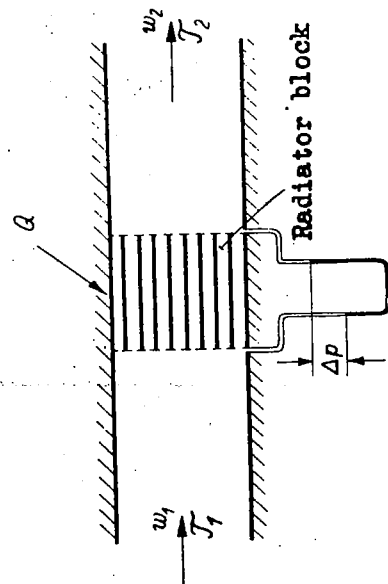
Figure 2.- Radiator block in closed
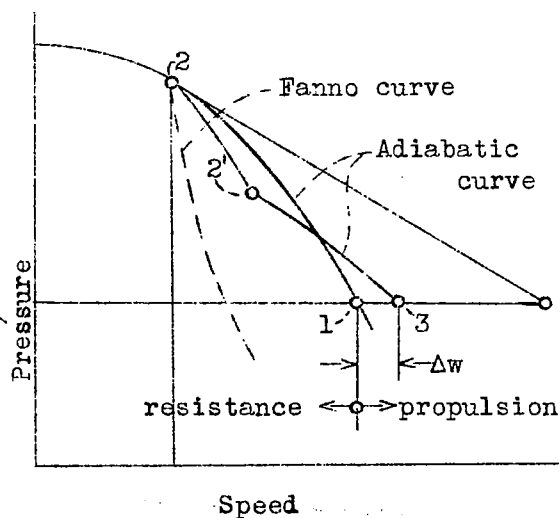   channel.
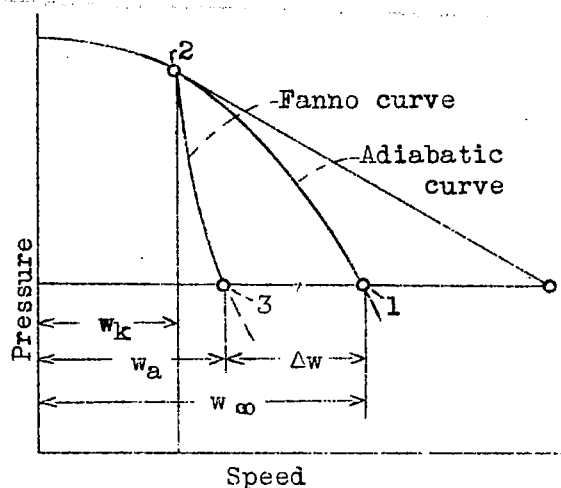
(a) Cold radiator, no nozzle.
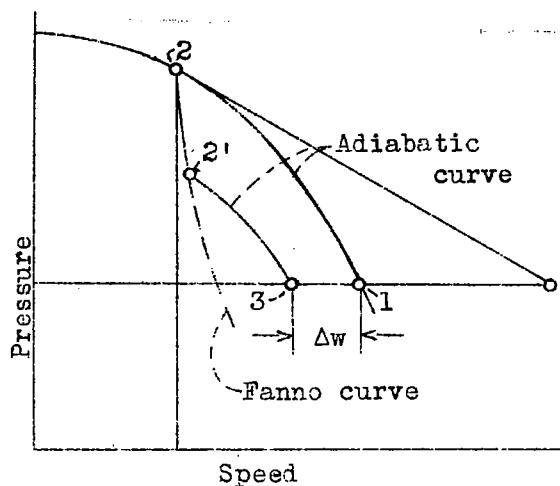
(b) Cold radiator, nozzle.
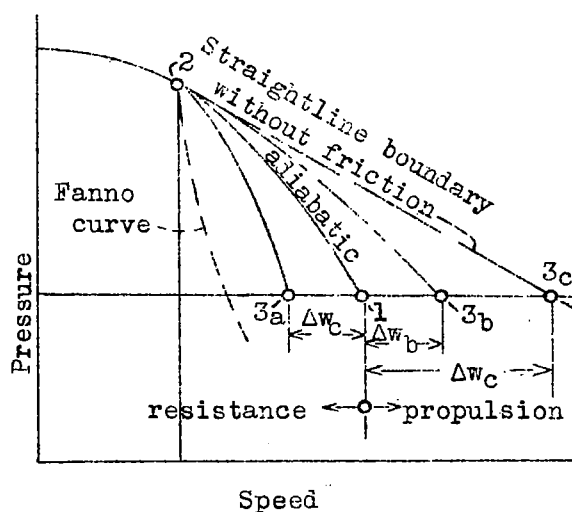
(c) Warm radiator, no nozzle.

(d) Warm radiator, nozzle.

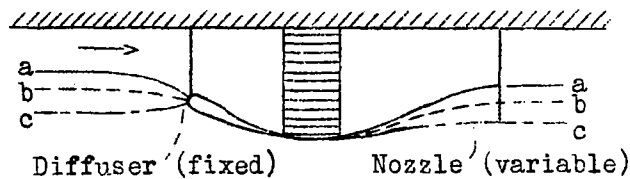Figure 6.- Working process of radiator in the pressure-speed diagram.

Figure 8.- Control of installed radiator thru adjustment of nozzle with fixed diffuser.
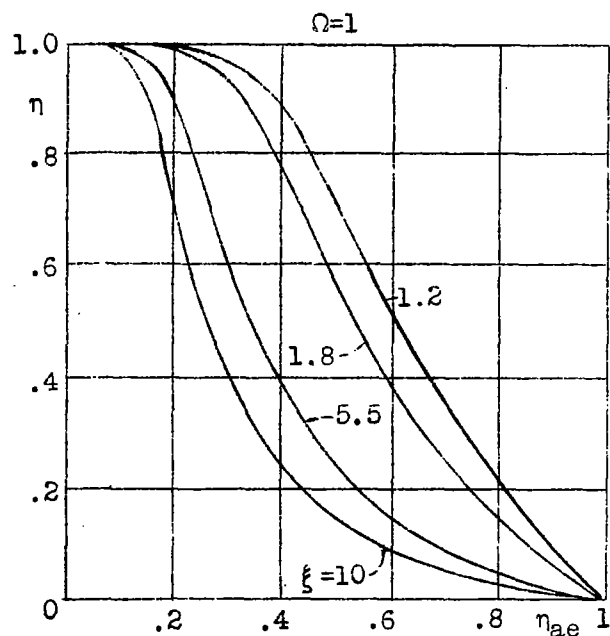

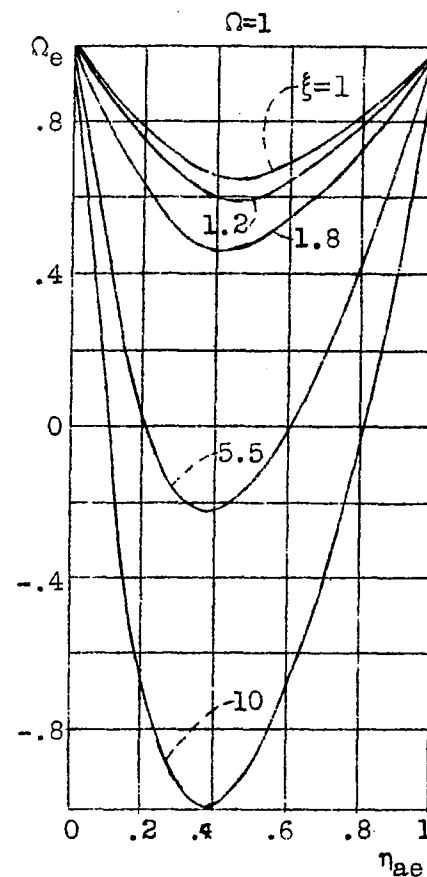Figure 9.- Heating of air in the radiator.

Figure 10.- Nondimensional resistance of radiators without nozzle.